MEASUREMENT AND MODELING OF BEARING DRAG IN IDLER ROLLERS

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ABSTRACT

Idler roller bearing drag plays a critical role when modeling web tension in a web line system. Lubrication, line speed, radial and axial loads on bearings can influence drag, which is directly reflected as a tension load on the web. An in-situ method has been devised for measuring idler roller bearing drag. Empirical models, fit to the data, provide bearing drag predictions under operating conditions. A relatively simple *Spin Down* test on an idler roller was used to predict the steady state drag from bearings as a function of line speed. Bearings were cleaned and lubricated with both oil and grease to test the contribution of lubricant to bearing drag.

NOMENCLATURE

B ....Linear torque model slope
I ....Roller rotational moment of inertia (kg-m^2)
t ....Time during roller spin down (s)
α ....Roller angular acceleration (rad/s^2)
τ ....Bearing torque (N-m)
τ_o ....Breakaway bearing torque (N-m)
τ_f ....Maximum bearing torque (N-m)
ω ....Roller angular velocity (rad/s)
ω_x ....Crossover angular velocity for nonlinear model (rad/s)

INTRODUCTION

The drag or torque, contributed by idler roller bearings, determines the tension required to turn the roller at web speed. Bearing drag must be taken into account when developing a realistic web tension control strategy for two reasons. First, sufficient web-to-roller traction must exist to drive a roller at web speed. Failure to do this will result in web slip and scratching of the web surface. A second reason is to avoid slack web conditions or excessive tensioning of delicate webs, caused by tension build up from multiple idler rollers in a tension zone.

Theoretical models of bearings depend on the physics of lubricated rolling and sliding contact interfaces between solids. Elastohydrodynamic lubrication (EHL) theory (1) has been successfully used to predict the thickness of lubricant in the contact area, taking into account the viscous nature of lubricants under pressure and elastic deformation of the contacting interfaces. Adequate lubrication is required in order to prevent premature failure of bearings due to direct metal-to-metal contact. A secondary goal of EHL theory has been to model bearing drag or torque as a function of speed and load conditions (2).

EHL requires knowledge of the precise geometric shape of bearing components, diametral clearance, elastic moduli, applied loads and angular race velocity. In addition, the viscosity of the lubricant as a function of shear rate, temperature and pressures approaching 3.5 GPa must be known. Bearing wear, metal fatigue, lubricant aging and contamination can have significant effects on bearing performance, making EHL modeling difficult for industrial applications.

RESEARCH GOAL

The goal of this research was to devise a relatively simple method for estimating bearing drag on an idler roller without removing it from the web line or having to depend heavily upon bearing geometry and exotic lubricant properties. An ideal, but impractical, approach to modeling bearings, is direct measurement of bearing torque as a function of the applied load and rotational velocity. Torque sensors, mounted between the machine side frame and the roller shaft, for "dead shaft" idler rollers, or between the side frame and bearing housing, for "live shaft" configurations would be expensive additions to a web line. In an off-line measurement, the roller must be removed from the web line and mounted in a instrumented test fixture. Not only would this be time consuming, but it might not accurately represent bearing drag when the roller was remounted in the web line. Finally, it is probably not necessary to continuously monitor bearing drag. Once a bearing and lubricant combination is evaluated, it should remain the same over the life of the bearing.

A test providing dynamic drag information without having to instrument any rollers, or remove them from the web line, would be ideal. A traditional test of bearing quality is to measure the time it takes a roller to coast to a stop from an initial velocity. The simplicity of this test, and its historical connection to bearing drag, prompted us to look for a relationship between the rotational speed of an idler roller during *Spin Down* and the torque exerted by the bearings. A test fixture was built using special rollers to simulate web line conditions of velocity and bearing load.

EXPERIMENTAL TEST METHOD

A set of five idler rollers was used to develop the measurement test method and bearing drag model. To facilitate cleaning and subsequent relubrication, all seals and shields were removed from the bearings. Before beginning a test sequence, all bearings were cleaned with solvent, and re-lubricated with the test lubricant. For all tests, the lubrication was added to only half of the bearing balls. A half hour run-in period, in the *Torque* configuration described below, was used to distribute the lubricant throughout the bearing. Table 1 contains bearing data, measured roller mass and rotational moment of inertia for each of the test rollers. Moments of inertia were calculated from roller geometry and density of materials.

Two representative lubricants were chosen for low drag idler roller lubrication. One was a standard detergent motor oil (3), and the other a low viscosity, pourable grease (4). While the oil was generally well behaved, some slinging of grease from the bearings was observed. Noncontacting grease shields, which were removed from the bearings, would have been sufficient to retain either of the two lubricants used.

In the first of two test configurations used, called a *Torque* test, the roller shell was held fixed at its outside diameter by a string anchored to a strain gage load cell (5) as shown in Fig. 1. A motor drove the roller shaft at different speeds, providing bearing dynamics similar to a rotating roller in a web line. Tension, or normal load on the bearings, was simulated by placing weights on the roller. These loads were balanced to prevent any static torque on the bearing. A Labview® (6) program automatically ramped the motor speed from zero to the maximum test value, collecting strain gage data during a test. Angular velocities ranged from 0 to 200 radians per second for testing radial loads from 21 N to 306 N.

A second configuration, called a *Spin Down* test, measured the roller surface velocity as it coasted to a stop from an initial rotational speed. Velocity was measured using a non-contacting laser surface velocimeter (LSV) (7) as shown in Fig. 2. A *Spin Down* test was initiated by first turning the shaft at maximum test speed and allowing bearing drag to bring the roller up to shaft speed. The shaft was dynamic braked by the motor, after which the roller coasted freely to a stop. The LSV measured surface velocities down to 30 m/min without contacting the roller.

Torque as a function of rotational velocity and load are shown in Fig. 3 for oiled and greased bearings. The experimental noise in this measurement is due mainly to mechanical vibration of the load cell which was velocity and load dependent. Increasing rotational velocity usually resulted in higher bearing drag. Torque-velocity curves for both oil and grease start out with similar slopes, but at velocities greater than about 100 rad/s greased bearing torque flattens.

Roller rotational velocity as a function of time is shown in Fig. 4 for oiled and greased bearings. The speed vs. time relationship did not depend on the initial roller speed used in the spin down test.

LINEAR TORQUE MODEL

Figure 3 suggests a linear model for torque as a function of rotational velocity

$$\tau(\omega) = \tau_0 + B \cdot \omega \tag{1}$$

where τ_0 is an initial breakaway torque and B is a constant. Using equation (1) and the following relationship between torque and angular acceleration, $\tau(\omega) = I \cdot \alpha$, a differential equation can be set up and solved for time as a function w of ω.

$$\tau_0 + B \cdot \omega = I \cdot \frac{\partial \omega}{\partial t} \tag{2}$$

$$t(\omega) = \frac{I}{B} \cdot \ln\left[\frac{\tau_0 + B \cdot \omega}{\tau_0}\right] \tag{3}$$

Because $\omega = 0$ implies time $t = 0$ in (3), it was necessary to reverse the time axis in Figure 4 for analysis. The time axis was also shifted to compensate for the absence of data at roller surface speeds below 30 m/min. By fitting equation (3) to spin down data, values for τ_0 and B were obtained. Figure 5 shows the torque curves predicted from *spin downs* of oiled and greased bearings. In both cases, the predicted torque fit reasonably well at low velocities but diverged above 100 rad/s. For many bearings, an improved torque model that could flatten out at higher velocities was required.

NONLINEAR TORQUE MODEL

A nonlinear torque function of the following form was chosen in an attempt to predict the torque vs. velocity curve more accurately.

$$\tau(\omega) = \frac{\tau_0 \cdot \omega_x + \tau_f \cdot \omega}{\omega_x + \omega} \tag{4}$$

Again τ_0 is an initial breakaway torque, τ_f is the final maximum torque achieved at $\omega = \infty$. The crossover velocity, ω_x, controls the transition between initial and final torque values. If ω_x is much larger than the highest roller velocity of interest, the torque curve will appear very similar to the linear model given by (1). Equation (4) was chosen for the following two reasons. First, it was well behaved at extreme values of ω and had physically meaningful parameters. Second, it resulted in a differential equation, similar to (2), that could be solved to give time as a function of ω.

$$t(\omega) = I \cdot \left\{\frac{\omega}{\tau_f} + \frac{\omega_x \cdot \left(\tau_f - \tau_0\right)}{\tau_f{}^2} \cdot \ln\left[\frac{\tau_0 \cdot \omega_x + \tau_f \cdot \omega}{\tau_0 \cdot \omega_x}\right]\right\} \tag{5}$$

As before, τ_0, τ_f and ω_x were obtained by fitting equation (5) to the roller *Spin Down* data. Model fits were indistinguishable from experimental *Spin Down* data for almost all bearing configurations tested. Predicted torque curves for oiled and greased bearings are shown in Fig. 6. The nonlinear model predicted oiled bearings better than the linear model, but a small amount of improvement was observed for greased bearings. In no case did the nonlinear model perform worse than the linear model. Both models gave excellent results at lower angular velocities with similar predicted values for the breakaway torque, τ_0.

Failure to model greased bearings at higher velocities, while frustrating, was not entirely surprising. Unlike the oil, the grease used was shear thinning and would be expected to behave quite differently during the short time scale of the spin down test vs. the steady state torque test.

BEARING DRAG AS FUNCTION OF APPLIED LOAD

Bearing load was expected to have an effect on the torque vs. velocity curve. Loads ranging from 41 N to 306 N were applied to Roller 2 (Table 1) with oiled bearings resulting in the torque curves shown in Figure 7. Surprisingly little load effect was observed with the exception of the heaviest 306 N load. Figure 8 shows the torque curves resulting from 23 N to 219 N loads applied to Roller 2 (Table 1) with greased bearings. In both cases, the torque predicted from spin down data (solid curves in Figures 7 and 8) fit the experimental data reasonably well for velocities below 100 rad/s. At higher velocities, the predicted torque was too large for oiled bearings at high load and greased bearings at low load. Suspecting bearing race slip as a possibility for divergence at high velocity, a proximity sensor (8) was set up to measure ball speed independent of the race speed. Within experimental error, the ball speed was consistent with rolling without race slip except for the highest load and velocity conditions.

Non-linear model fits to both *Spin Down* and *Torque* data for all five rollers are shown in Table 2. For the *Spin Down* test, the load applied to the roller is contributed by the roller's mass. The *Torque* test includes both the roller mass and external load on the roller. Model parameters for a best least squares fit to the *Spin Down* curve as well as the torque predicted at 100 and 200 rad/s roller velocity are shown in unshaded areas of the table. Initial and final torque values from the model fit are in rows labeled **T0** and **T1** respectively, while the crossover velocities are in rows labeled **W**. Shaded areas of the table contain parametric fits to the measured torque data. Spin Down model deviations are shown as percentages below the actual torque observed for 100 and 200 rad/s. While deviations as high as 83% were observed, the predictions from the *Spin Down* test were reasonably good considering the simple and direct modeling technique employed.

In an attempt to improve the model fit, the model was corrected by a scaling function derived from the applied load. This scaling function depended on the logarithm of the load and was quite different for oiled and greased bearings. Scaled model predictions, however, proved little better fit to the torque data and the added complexity of this approach was deemed not worth the effort. It was hoped that EHL theory (1) might be used to provide a relatively simple load correction factor to the *Spin Down* torque model. The unpredictable nature of the measured torque/load dependence and its obvious variation with lubricant, made a simple EHL correction factor seem unlikely.

More research will be needed to combine the strengths of theoretical and empirical bearing models.

DRAG AS A FUNCTION OF BEARING SIZE

Measured torque values in table 2 show a general increase with bearing size. This was more evident in the greased bearings than the oiled ones. Figure 9 shows the bearing size dependence at a angular velocity of 100 rad/s and a narrow load range around 135 N. With one exception, the greased bearings performed better midrange bearings but no better for the largest bearing. Given a factor of five increase in torque with bearing size, it makes sense to choose the smallest bearing that will handle the required load conditions. No correction to the *Spin Down* model for bearing size was necessary as the model naturally scaled with the size of the bearings measured.

CONCLUSIONS

An empirical relationship was developed between a roller *Spin Down* test and bearing torque as a function of velocity. Good predictability, for oiled and greased bearings, was observed for roller velocities up to 100 rad/s. This angular velocity limit represents a web velocity of 381 m/min (1250 ft/min), covering the velocity range of most plastic film products. Bearing load contributed a weak logarithmic increase to the drag and could be neglected from *Spin Down* model predictions without incurring much error. A marked increase was observed in bearing torque as a function of bearing size. *The Spin Down* model takes this factor into account naturally.

From the low bearing torque observed for both oiled and greased bearings, high drag attributed to idler rollers probably comes from sources such as tight seals, contaminated or worn out bearings.

ACKNOWLEDGMENTS

The authors would like to thank Shabaka McKey for his contributions to making the bearing test fixture and running some of the first experiments. We would also like to thank Webex Inc. for providing support and consultation in choosing bearings and manufacturing the test rollers.

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4. Lubriko M-21 grease, Master Lubricants Co., Philadelphia PA.
5. LCF-5, Omega Engineering, Inc., Stamford, CT.
6. National Instruments Corp., Austin, TX.
7. LSV-040, Polytec GmbH, Waldbronn, Germany.
8. FS-17, Keyence Corp., Osaka, Japan.

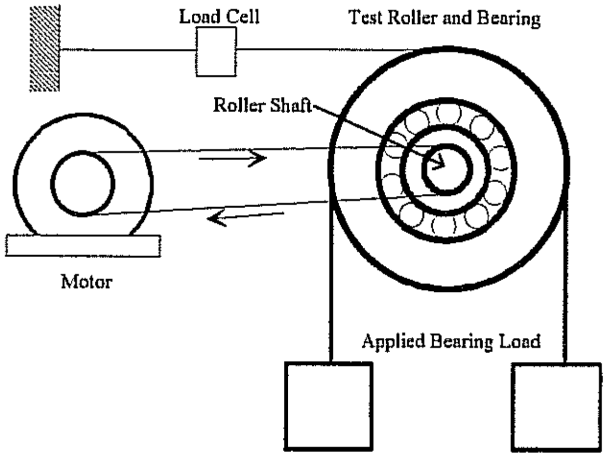

Fig. 1 Bearing torque measurement test configuration.

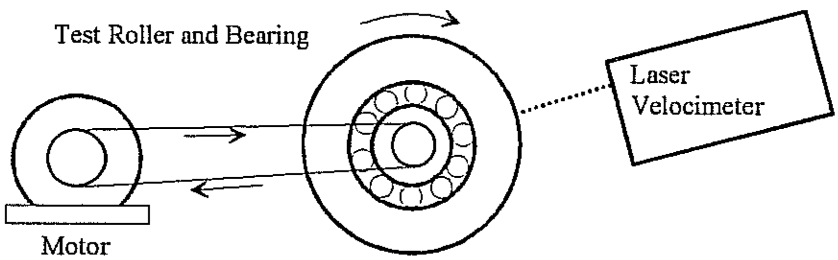

Fig. 2 Roller spin down test configuration

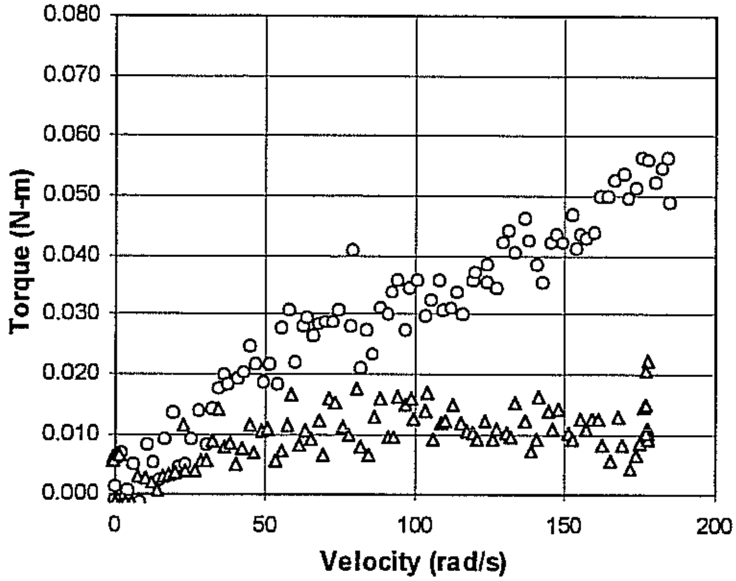

Fig. 3 Torque as a function of angular velocity for roller 2. Experimental data shown for oiled (o) and greased (Δ) bearings.

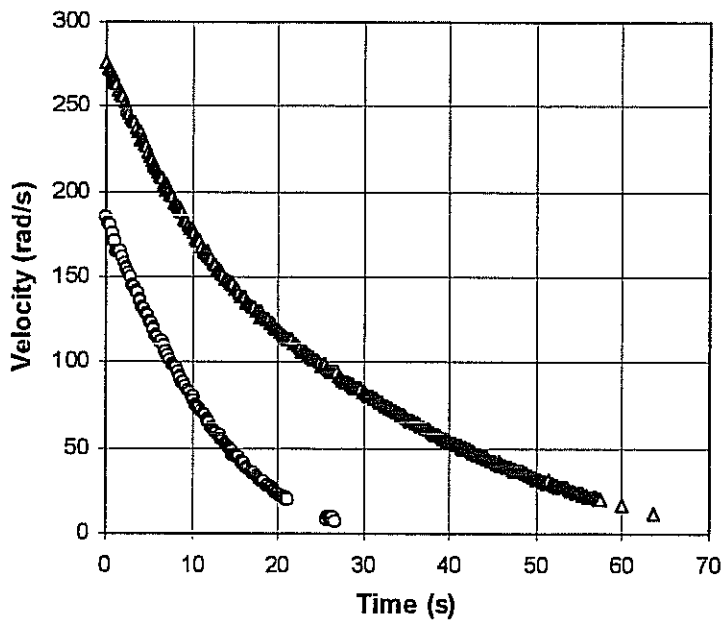

Fig. 4 Angular velocity as a function of time for roller 2 spin down. Experimental data shown for oiled (o) and greased (Δ) bearings.

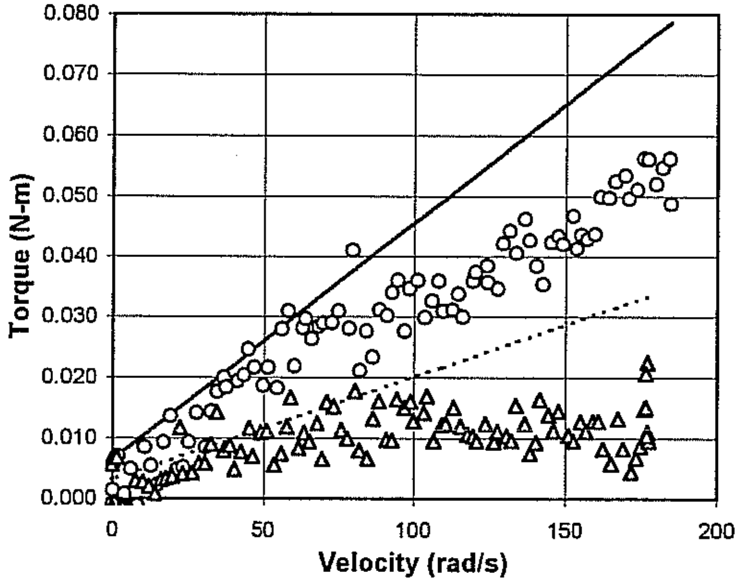

Fig. 5 Torque data as a function of angular velocity as in Figure 3. Linear fits to data from roller spin down are shown for oiled (solid) and greased bearings (dotted).

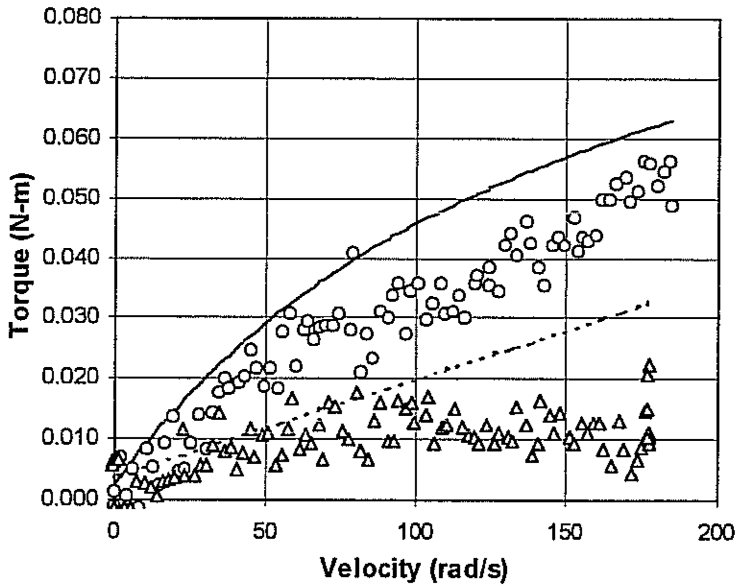

Fig. 6 Torque data as a function of angular velocity as in Figure 3. Nonlinear fits to data from roller spin down are shown for oiled (solid) and greased bearings (dotted).

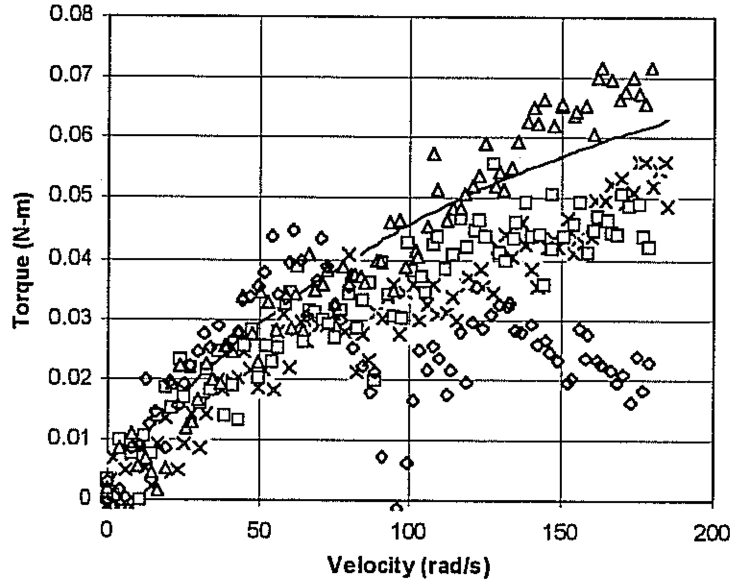

Fig. 7 Torque as a function of angular velocity under different applied loads for oiled bearings in roller 2. Roller loads are 41 N (x), 130 N ( ), 219 N (Δ) and 306 N ( ). Solid curve is nonlinear fit from roller spin down as in Fig. 6.

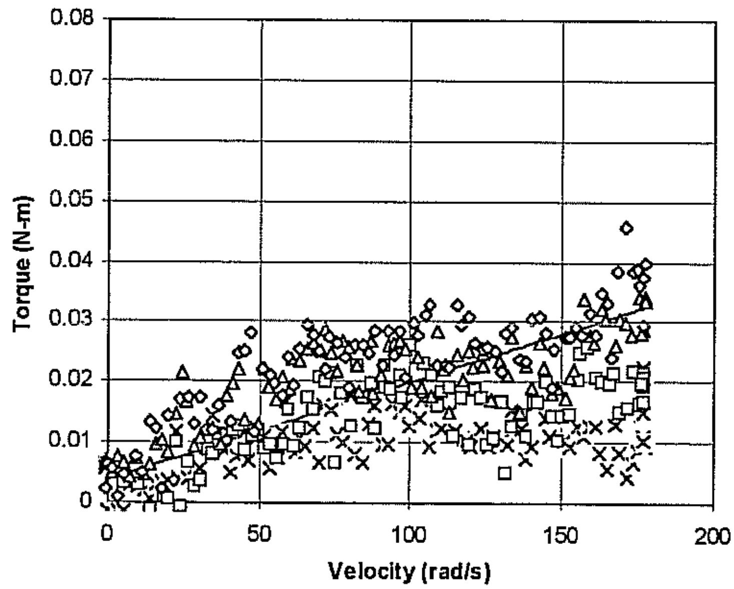

Fig. 8 Torque as a function of angular velocity under different applied loads for greased bearings in roller 2. Roller loads are 23 N (x), 41 N ( ), 130 N (Δ) and 219 N ( ). Solid curve is nonlinear fit from roller spin down as in Fig. 6.

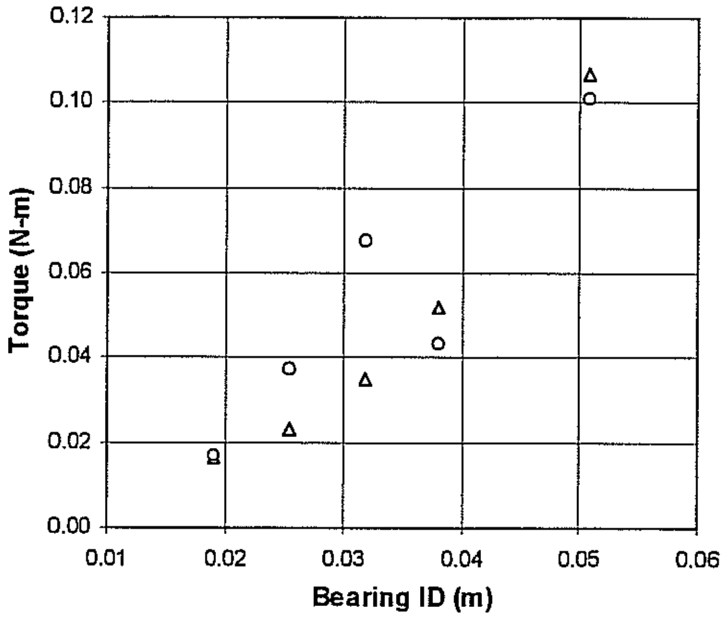

Fig. 9 Torque as a function of bearing ID for oiled (o) and greased (Δ) bearings. All data at 100 rad/s radial velocity and loads ranging from 130 to 149 N.

Roller number	1	2	3	4	5
Shaft diameter (m)	0.01905	0.02540	0.03175	0.03810	0.05080
Roller mass (kg)	2.40	2.34	3.07	3.47	4.37
Rot. moment (kg-m^2)	0.005021	0.004992	0.005034	0.005072	0.004494
Inner-race dia. (m)	0.025588	0.031382	0.042614	0.047300	0.063213
Outer-race dia. (m)	0.041486	0.047285	0.064867	0.072728	0.091816
Ball dia. (m)	0.007938	0.007938	0.011113	0.012700	0.014288
Number of balls	8	9	9	9	10
Inner-groove rad. (m)	0.004140	0.004140	0.005791	0.006604	0.007442
Outer-groove rad. (m)	0.004128	0.004128	0.005779	0.006604	0.007430

Table 1 Test roller and bearing properties. Aluminum roller shells were 0.1016 m in width and 0.127 m in diameter.

		Oiled Bearings				Greased Bearings			
		Spin	Torque			Spin	Torque		
ROLLER 1	Load (N)	23.6	41.4	130.3	219.3	23.6	41.4	130.3	219.3
	T0 (N-cm)	0.11	-0.28	-0.47	0.20	0.17	-0.02	0.29	-0.35
	T1 (N-cm)	4.95	3.60	5.01	5.89	11.08	1.67	2.39	1.89
	W (rad/s)	224	85	155	105	678	23	55	15
	Torque @	Predict	1.82	1.68	2.97	Predict	1.36	1.64	1.60
	100 (rad/s)	1.61	-12%	-4%	-46%	1.57	16%	-4%	-2%
	Torque @	Predict	2.44	2.62	3.93	Predict	1.50	1.94	1.73
	200 (rad/s)	2.40	-2%	-9%	-39%	2.65	77%	37%	53%
ROLLER 2	Load (N)	22.9	40.7	129.7	218.6	22.9	40.7	129.7	218.6
	T0 (N-cm)	0.25	0.13	0.17	0.17	0.38	-0.66	0.45	0.19
	T1 (N-cm)	11.60	13.39	7.63	22.55	2.6E+5	2.45	3.48	4.40
	W (rad/s)	162	306	108	403	1.6E+7	37	62	70
	Torque @	Predict	3.40	3.75	4.62	Predict	1.61	2.32	2.67
	100 (rad/s)	4.58	35%	22%	-1%	1.99	24%	-14%	-25%
	Torque @	Predict	5.37	5.01	7.60	Predict	1.97	2.76	3.31
	200 (rad/s)	6.52	21%	30%	-14%	3.61	83%	31%	9%
ROLLER 3	Load (N)	30.1	47.9	136.9	225.8	30.1	47.9	136.9	225.8
	T0 (N-cm)	0.02	0.21	0.37	0.46	0.76	-0.30	-0.35	-0.35
	T1 (N-cm)	5.65	8.23	15.22	14.24	10.55	3.37	7.45	7.45
	W (rad/s)	51	55	132	112	292	18	103	103
	Torque @	Predict	5.38	6.78	6.96	Predict	2.82	3.48	3.48
	100 (rad/s)	3.75	-30%	-45%	-46%	3.26	16%	-6%	-6%
	Torque @	Predict	6.50	9.32	9.29	Predict	3.07	4.79	4.79
	200 (rad/s)	4.51	-31%	-52%	-52%	4.74	54%	-1%	-1%
ROLLER 4	Load (N)	34.0	51.8	140.8	229.7	34.0	51.8	140.8	229.7
	T0 (N-cm)	0.11	0.88	0.87	1.28	0.42	-0.24	-0.38	0.06
	T1 (N-cm)	48.66	17.04	1.6E+4	7.3E+4	9.25	4.35	6.02	9.57
	W (rad/s)	2162	767	4.5E+5	1.9E+6	84	13	14	46
	Torque @	Predict	2.74	4.36	5.07	Predict	3.82	5.21	6.57
	100 (rad/s)	2.26	-18%	-48%	-55%	5.23	37%	0%	-20%
	Torque @	Predict	4.22	7.84	8.86	Predict	4.07	5.59	7.79
	200 (rad/s)	4.22	0%	-46%	-52%	6.65	63%	19%	-15%
ROLLER 5	Load (N)	42.5	60.3	149.2	238.2	42.5	60.3	149.2	238.2
	T0 (N-cm)	0.06	-0.22	1.42	0.72	0.00	0.30	1.09	1.41
	T1 (N-cm)	4.85	9.19	25.74	30.21	8.91	10.91	121.72	186.77
	W (rad/s)	31	70	181	201	65	51	1158	1608
	Torque @	Predict	5.31	10.09	10.51	Predict	7.32	10.68	12.26
	100 (rad/s)	3.71	-30%	-63%	-65%	5.41	-26%	-49%	-56%
	Torque @	Predict	6.75	14.20	15.41	Predict	8.75	18.86	21.91
	200 (rad/s)	4.20	-38%	-70%	-73%	6.73	-23%	-64%	-69%

Table 2 Results from *Spin Down* and *Torque* tests on all five rollers with oiled and greased bearings. See text for details.

Question - Did you consider bearing inertia in the roller inertia?

Answer - Yes

Question - So you did calculate that in there.

Answer - It was small.

Question - Secondly, the Oil grease relationship is just the opposite of what I've sometimes seen. What was the characteristic of the grease?

Answer - This grease was a pourable grease. Its not like anything that comes out of a can. This comes out of a tube and is very loose.

Comment: Typical Seal grease bearing has a straight line torque characteristic.

Answer - Yes, we did measure a heavier base of lithium grease. Our roller speed measurement system did not handle lower velocities very well and we couldn't get a long enough spin down to do a good spin down test. The morale of the story is to use a good roller measurement system.

Question - Was it oil filled or a few drops of oil?

Answer - We filled up the bearing about half full. We did sling a little oil so the oil did not necessarily stay there.

Question - Please review how you choose your τ_0, τ_f on the plots from the nonlinear model. Was that just by inspection of the plots or how did you decide what figures to put in there?

Answer - We used EXCEL. We set up a spreadsheet with solver which iterates those parameters until you have a best least squares fit to the spin down data.

Question - So you fit all three parameters empirically through the solver for data.

Answer - Yes, then we used those through integration to predict torque. On one hand it is an imperical model for torque, but it is through solving the integration you get to spin down.

Question - Do you have any plans to extend this to a live shaft roller, and possibly to seals and packing?

Answer - The on-line technique will work for any roller. You just measure the spin down and fit the model to it. Seals and packing only increase roller drag. The spin down testing equipment has to be good. If you have a spin down that is very fast you are going to have a hard time measuring velocity as a function of time.